Active Alignment and Vibration Control System for Large Airborne Optical System

David A. Kienholz
CSA Engineering, Inc.
2565 Leghorn Street
Mountain View, CA

Presented at SPIE
Smart Structures and Materials Conference
San Diego, CA
March, 2000

Active Alignment and Vibration Control System for a Large Airborne Optical System

David A. Kienholz
CSA Engineering, Inc.
Mountain View, CA

ABSTRACT

Airborne optical or electro-optical systems may be too large for all elements to be mounted on a single integrating structure, other than the aircraft fuselage itself. An active system must then be used to maintain the required alignment between elements. However the various smaller integrating structures (benches) must still be isolated from high-frequency airframe disturbances that could excite resonances outside the bandwidth of the alignment control system. The combined active alignment and vibration isolation functions must be performed by flight-weight components, which may have to operate in vacuum. A testbed system developed for the Air Force Airborne Laser program is described. The payload, a full-scale 1650-lb simulated bench, is mounted in six degrees-of-freedom to a vibrating platform by a set of isolator-actuators. The mounts utilize a combination of pneumatics and magnetics to perform the dual functions of low-frequency alignment and high-frequency isolation. Test results are given and future directions for development are described.

Keywords: vibration isolation, active systems, optical alignment, vibration control

1. INTRODUCTION

Vibration is always a design consideration in airborne (or any vehicle-borne) optical system. Besides affecting external performance such as beam pointing accuracy or line-of-sight jitter, it affects alignment between internal components. The alignment problem is particularly significant in large systems where it may not be possible to mount all the major components on a single integrating structure, other than the aircraft fuselage itself. Being strength-driven rather than stiffness-driven, a typical fuselage structure is not well suited to the task of passive alignment maintenance; airplanes do not make good optical benches.

Ground-based optical systems up to a few thousand pounds are typically mounted on passive isolation mounts to reduce the effects of ground motion. Primarily, they reduce vibration of the integrating structure by reducing the excitation in the frequency ranges around its own resonances. The mounts are simply soft springs that produce isolation at frequencies above the induced suspension natural frequencies. The latter are typically on the order of 2-5 Hz. Applying this passive technique to airborne systems immediately raises two problems. Floor vibration levels in large aircraft are typically 1.5-3 orders of magnitude higher than in ground facilities, thus requiring greater levels of isolation. Secondly, an airborne system is subjected to very low frequency floor accelerations caused by aircraft maneuvering. This would cause large displacements between the isolated payload and the fuselage, or between payloads on separate isolation systems.

It is to these problems that the present development has been addressed. The objective is a passive isolation system with performance superior to existing laboratory systems but capable of operating at much larger amplitudes. It is then augmented by an active subsystem capable of dealing with the very low frequency inertial loads imposed on the payload by aircraft maneuvering. The result is called AS/VIS: Airborne Stabilization and Vibration Isolation System. A testbed demonstration version is described in this paper.

2. SYSTEM DESCRIPTION

An early design decision was to use pneumatics as the basis of the passive system. Air springs allow the dynamic restoring stiffness to be decoupled from the static, weight-induced sag of the payload. Vertical suspension frequency becomes independent of payload weight.[1] Conventional air springs use an elastomeric air bag or rolling diaphragm to contain a quantity of pressurized air which is compressed or expanded by vertical motion of the payload. Horizontal restraint is provided by the flexural stiffness of the air bag itself or by pendulum action induced by the mechanism

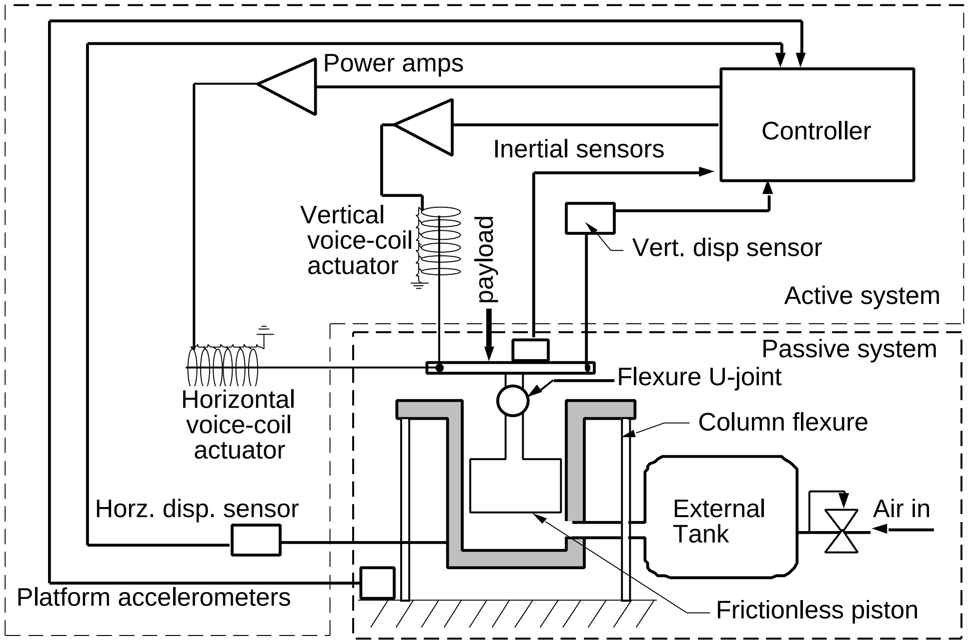


Figure 1. Operating principle of pneumatic-magnetic active isolation system.

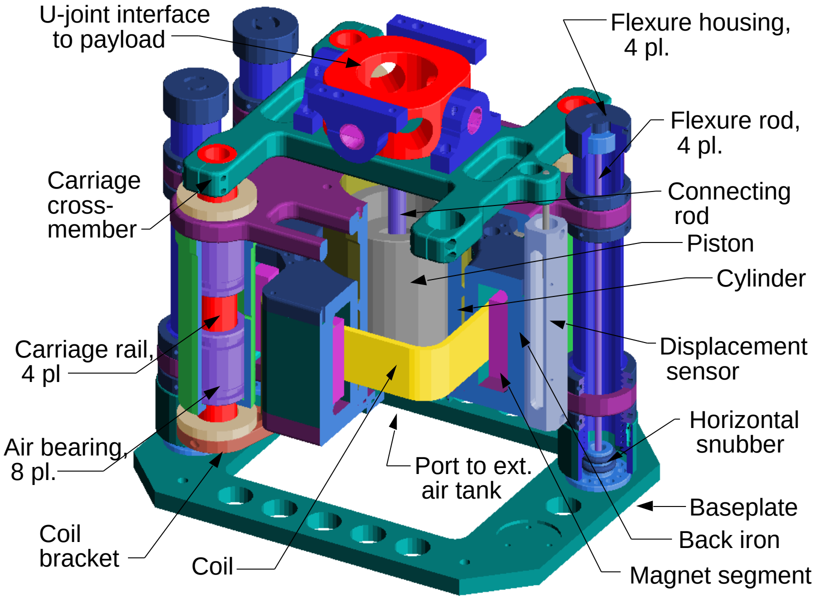
connecting the air spring and payload. Commonly used in lab systems, such approaches produce a lower limit to suspension frequency of about 2 Hz due to stiffness of the air bag itself. This and the inherent damping and hysteresis of the elastomer set performance limits for conventional pneumatic isolators.

AS/VIS applied a pneumatic technology developed originally for suspension systems used to simulate the zero-gravity conditions of orbit for ground vibration testing of spacecraft.[2] Detailed in following paragraphs, it allows arbitrarily low vertical stiffness with zero sag, long stroke, and zero friction. It also provides damping of vertical suspension resonances without compromising isolation at higher frequencies.

Requirements on the actuators led to the choice of electromagnetic voice coils. They had to be capable of relatively long stroke: over 1 inch. Force capacity had to be sufficient to control a payload of 1000-2000 lbs under floor acceleration levels as high as 0.3 G peak. A high degree of linearity and zero friction were required for alignment control accuracy. Bandwidth requirements were moderate: 10-20 Hz maximum. These requirements led to voice-coil actuators based on rare earth magnets with air bearing suspension of the moving coil.

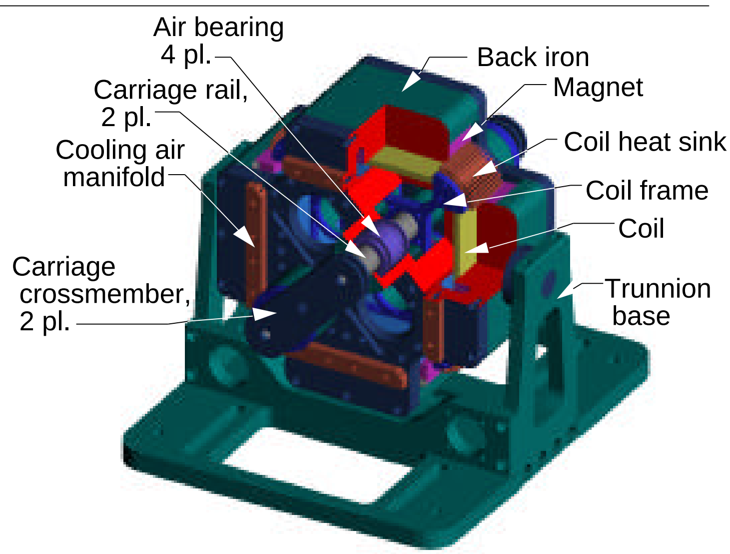
Figure 1 shows a simplified schematic of the system. The payload weight is supported on four identical pneumatic springs (vertical isolator/actuators in Figure 2). Each uses a special frictionless piston developed for the zero-g suspension noted earlier. Each air cylinder is plumbed through a large diameter line to an external accumulator tank sized to adjust the effective stiffness of the air spring. Each cylinder-tank combination is fed by a precision mechanical pressure regulator. A voice coil actuator and vertical displacement sensor operate in parallel with each air spring. The coil and magnet body of of the actuator surround the air cylinder for compactness. These elements comprise the frame of the device which is supported on a set of four slender vertical rods that serve as flexures, allowing high compliance in the horizontal direction. The air springs are sized for a vertical suspension frequency of about 0.7 Hz without active augmentation. A displacement feedback loop is used to add a small amount of active centering stiffness, raising the frequency to about 1.05 Hz with a payload of 410 lbs per device. The flexures are sized to place the horizontal suspension modes at about 1.2 Hz with this load. The vertically moving carriage of the device runs on eight journal air bearings as shown. Summary specifications for the vertical airmount isolator/actuators are given in Figure 2.

AS/VIS is based on a “rectangular” geometry. The entire payload weight is supported on four isolator/actuators having their air springs and actuators oriented vertically. They provide passive isolation in all six DOFs plus control authority over three of the six rigid-body modes of the payload. In addition, four additional voice coil actuators operate exert force on the payload, two in each horizontal direction to control the remaining three DOFs. The horizontal actuators (lower part of Figure 2) use the same voice coil design as the verticals but have much lighter carriages and air bearings since they support no payload weight.



Vertical airmount isolator / actuator

Piston bore:	4.0 in
Payload capacity (per mount) :	
at 30 psig	350 lbs
at 100 psig	1230 lbs
Vertical frequency (350-lb payload)	
active off, 2.0 gal. tank	0.66 Hz
active off, 10.0 gal. tank	0.32 Hz
active on, 24 lbf/in, 2.0 gal. tank	1.05 Hz
active on, 400 lbf/in, no ext. vol.	3.55 Hz
Horizontal frequency (350-lb payload)	
active off	1.26 Hz
active on (400 lbf/in)	3.10 Hz
Stroke (bumper to bumper)	
vertical	1.25 in
horizontal	1.00 in
Friction (% of payload):	< 0.01%
Actuator force constant	24.0 lbf/amp
Actuator force capacity	
intermittent	220 lbf
sustained	180 lbf
Weight	146 lbf



Horizontal actuator

Force capacity:	
intermittent	230 lbf
continuous	190 lbf
Force constant:	25.0 lbf/amp
Stroke (bumper to bumper)	1.1 in
Bandwidth (3 dB)	> 40 Hz
Carriage suspension stiffness	zero
Carriage friction	zero
Weight	96 lb

Figure 2. Vertical airmount isolator/actuator and horizontal actuator.

The magnetic design of the actuators is somewhat unconventional. The coils are square in axial view with rectangular magnets positioned along each of the four sides to drive flux radially inward through the coils. In the vertical actuators, the coils attach at their corners to “legs” that carry their actuation force to the moving carriage. In the horizontals, this function is performed by end frames to which the coils attach. In both cases, the arrangement allows the large, rare earth magnets to be flat slabs for reduced cost relative to the more common annular segments. Both verticals and horizontals have integral forced air cooling. A small amount of pressurized air from the bearings is directed over the coils in such a way as to entrain a greater volume of ambient air for cooling. The overall actuator design is aimed at high thrust to weight ratio at the expense of bandwidth. Disturbances from aircraft maneuvering and airframe global modes are generally confined to frequencies below about 10 Hz. Above this, the system is designed to act simply as a 1-Hz passive isolator and little or no actuation authority is needed.

A full-scale system-level demonstration testbed has been constructed using the isolators and actuators described above (Figure 3). As noted, it uses four vertical isolator/actuators and four horizontal actuators. The system is mounted on a 60 x 100-inch vibrating platform constructed with a steel internal skeleton and plywood face sheets. The platform rests on conventional airbag mounts to give a vertical suspension frequency of about 8 Hz. A servohydraulic actuator mounts to the floor under the platform and allows shaking it with a controlled spectrum in the vertical direction at levels up to about 1 g peak. The smaller photo in the lower left shows one of four airmount isolator/actuators and one of four horizontal actuators installed in the testbed.

The 1650-lb payload, the large black structure in the top photo of Figure 3, is a dummy structure constructed of welded steel tubing with sandwich shear panels on the sides, top, and bottom. Each panel has a core layer of viscoelastic material to damp the flexural modes of the structure. The intent is to make the payload behave essentially as a rigid mass without resonances anywhere close to the control band, thus allowing experiments to concentrate on the behavior of the isolation/stabilization system.

In addition to the relative displacement sensors built into the eight actuators, the testbed includes 16 channels of DC-coupled accelerometers. Eight are mounted on the vibrating platform and eight on the isolated payload. Signal conditioning for all sensors as well as power amplifiers and preamps for the actuators are mounted in the rack on the right in the upper photo in Figure 3. The console on the near end of the platform contains controls for the air bags under the platform, air bearings of the actuators, and cooling air for the actuators. Precision piston pressure regulators for the airmounts are mounted in the control rack.

The testbed was built primarily for investigating various active isolation and dynamic alignment control strategies utilizing pneumatic-magnetic isolator technology. It includes a VME-bus real time processor and analog interface boards. A deterministic real time operating system is used for control implementation, performance monitoring, and data logging. Thirty-two channels of A/D and eight channels of D/A conversion are supported. The inputs are usually the eight displacement sensors and sixteen accelerometers. Outputs go to the command inputs for the eight actuators. The electronics are arranged such that control loops can be rapidly reconfigured and analog sensor signals can be easily tapped into for diagnostic testing. Various all-analog modes are also possible for hardware checkout.

3. TEST RESULTS

The demonstration hardware has been tested extensively at both the component and system level. This section presents some typical results.

An early test was directed at characterizing the regulated air springs of the airmounts. The cylinder and external tank of each airmount are fed by a precision mechanical regulator which tries to keep the pressure constant as the piston moves vertically in the cylinder. To the extent it can do this, it will suppress the stiffness of the air spring. The regulator is a commercial unit whose bandwidth depends on the volume of the accumulator tank. At sufficiently low frequency, it reduces the air spring stiffness to essentially zero. A test was performed to measure the frequency-dependent stiffness of the regulated air spring in one of the airmounts. A strain gauge pressure transducer was used to measure piston pressure and the integral LVDT displacement sensor was used to sense piston displacement. An airmount was configured to support a rigid mass and the integral voice coil actuator was used to excite the carriage with a band-limited random waveform. A digital Fourier analyzer was then used to measure the frequency response between vertical carriage displacement as input and piston pressure as output. Multiplying this function by the piston area gives the complex stiffness of the air spring. Figure 4 shows the result for several mass loads. As expected, the high-pass break frequency is independent of pressure. It can also be shown to vary inversely with accumulator

Figure 3. System-level testbed (top). Vertical isolator/actuator and horizontal actuator (bottom left). Hydraulic actuator for shaking platform (lower right).

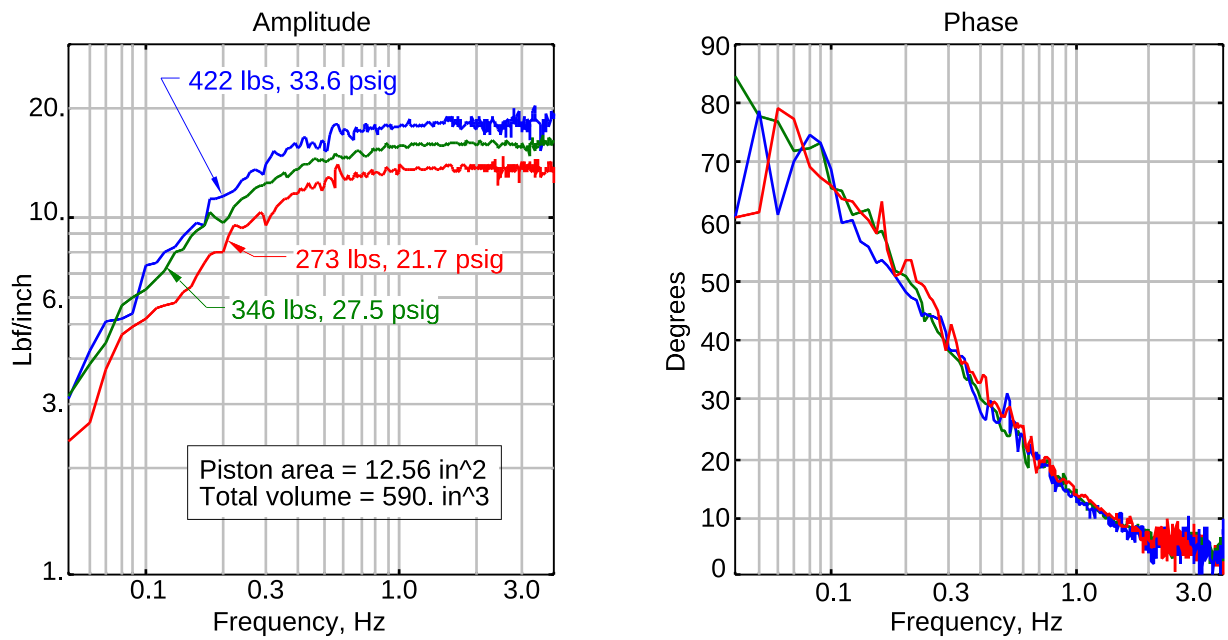


Figure 4. Measured complex stiffness of regulated air spring.

tank volume. This experimentally obtained stiffness of the regulated air spring is essential to accurate simulation and modeling of the system. The high-frequency asymptote in the complex stiffness curve is herein referred to as the "stiffness" of the air spring.

A second test of an individual airmount was performed to determine the isolation performance in a way that was independent of the dynamics of any large payload or shaker platform. Figure 5 shows the test setup and results. One isolator/actuator unit was used as a shaker to impose a controlled vertical base motion on a second unit supporting a rigid payload. The airmounts proved quite adept at this because the low-stiffness air spring of the bottom unit could "float" the unit under test and its payload while the actuator provided sufficient dynamic force to impose a controlled AC base acceleration. Results in terms of acceleration transmissibility are shown in Figure 5 for several different sizes of accumulator tank. Isolation corner frequencies as low as 0.5 Hz were readily obtained, resulting in 40 dB of isolation at about 4 Hz. Even with a 2 gallon tank (the size used with each airmount in the system-level testbed) the corner frequency is only about 1 Hz and 40 dB is achieved at about 8 Hz.

The demonstration testbed of Figure 3 has been used to assess system-level isolation performance. Using a hydraulic shaker under the platform to produce vertical vibration, transmissibility was measured between it and the payload in the vertical direction. In this context, transmissibility is simply the frequency response between a vertical accelerometer on the platform as input and a vertical accelerometer on the payload as response. This is not the same as the classical single-DOF definition because the platform is not rigid and the resulting FRF will depend on which platform accelerometer is picked as the input quantity. Nonetheless, the results are a reasonable indication of performance. Figure 6 shows the tranmissibility for two different choices of input and output locations. One pair, denoted FZ1 and BZ1, is on the forward ends of the platform and payload (away from the camera in the top photo in Figure 3) and the other pair, FZ4 and BZ4, is on the aft end. For this test, each airmount had its displacement sensor and actuator configured in a simple, uncompensated displacement loop with a gain of approximately 25 lbf/inch. An isolation break frequency of about 1 Hz is apparent. The rolloff rate is about 40 dB per decade with resonant peaking of about 10 dB. The latter quantity could readily be reduced even further by appropriate velocity feedback to the actuators at low frequency. Figure 7 shows time histories of typical acceleration on the platform and payload. The RMS level on the platform was about 0.1 G rms, typical of an aft location of a large aircraft flying in heavy turbulence. RMS acceleration on the payload is reduced by over 40 dB by the isolation system.

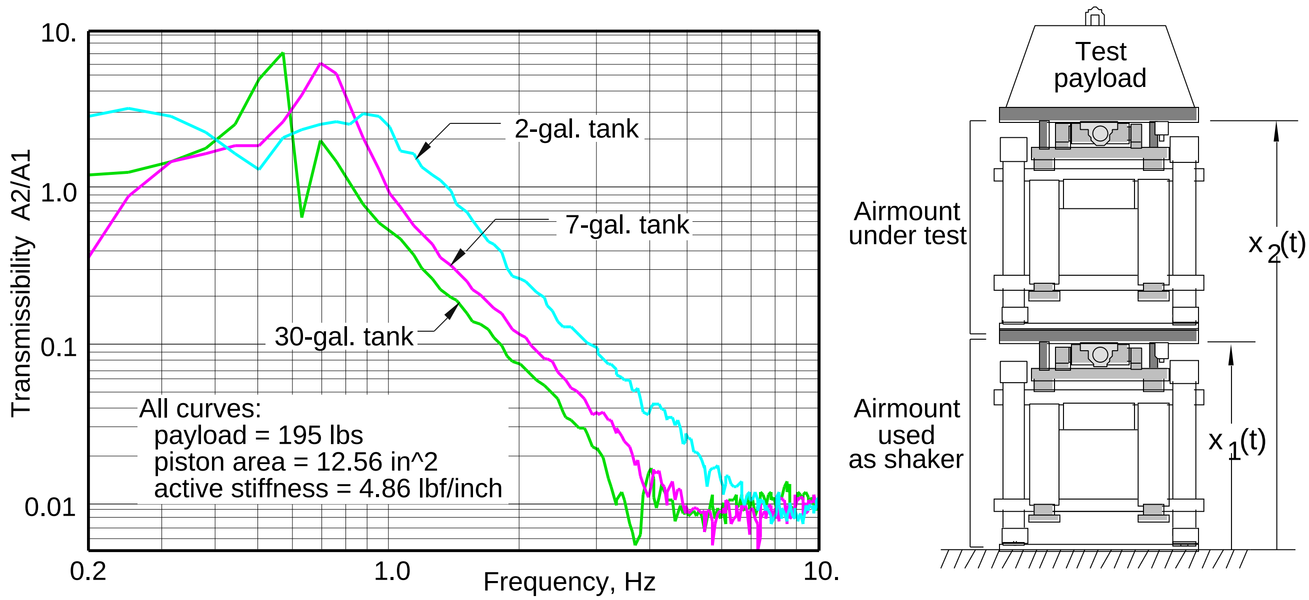


Figure 5. Transmissibility test of single airmount isolator/actuator.

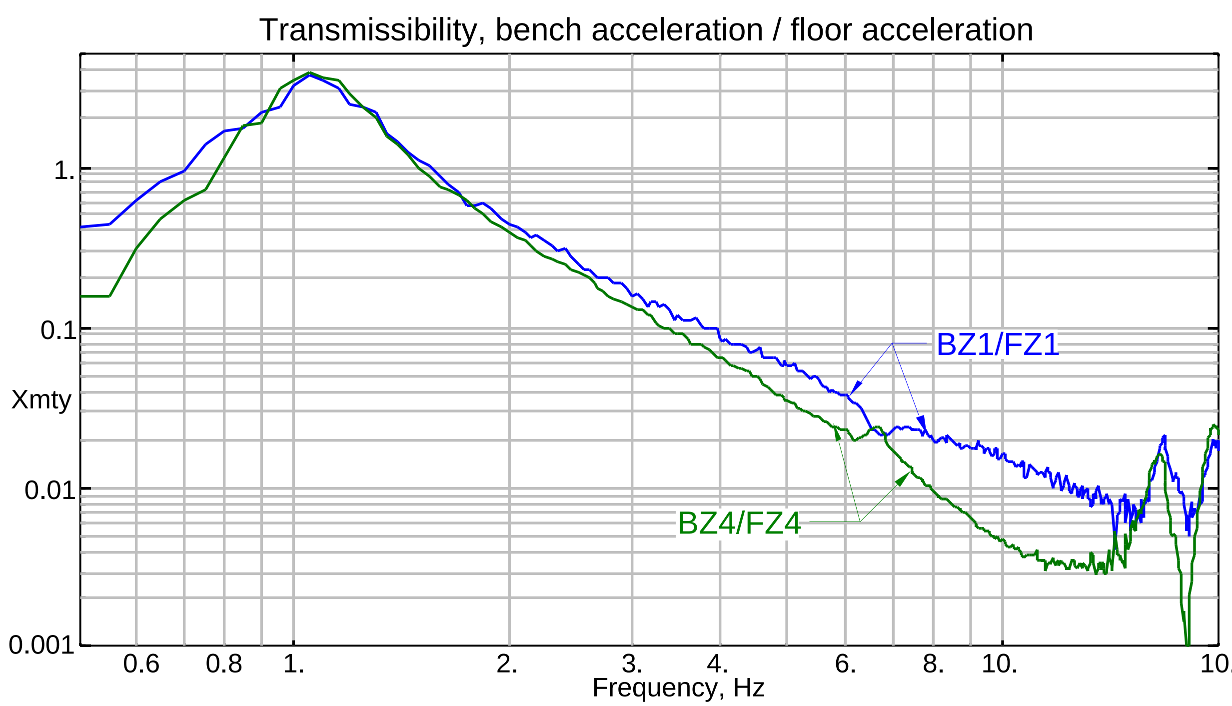


Figure 6. Measured system-level acceleration transmissibility

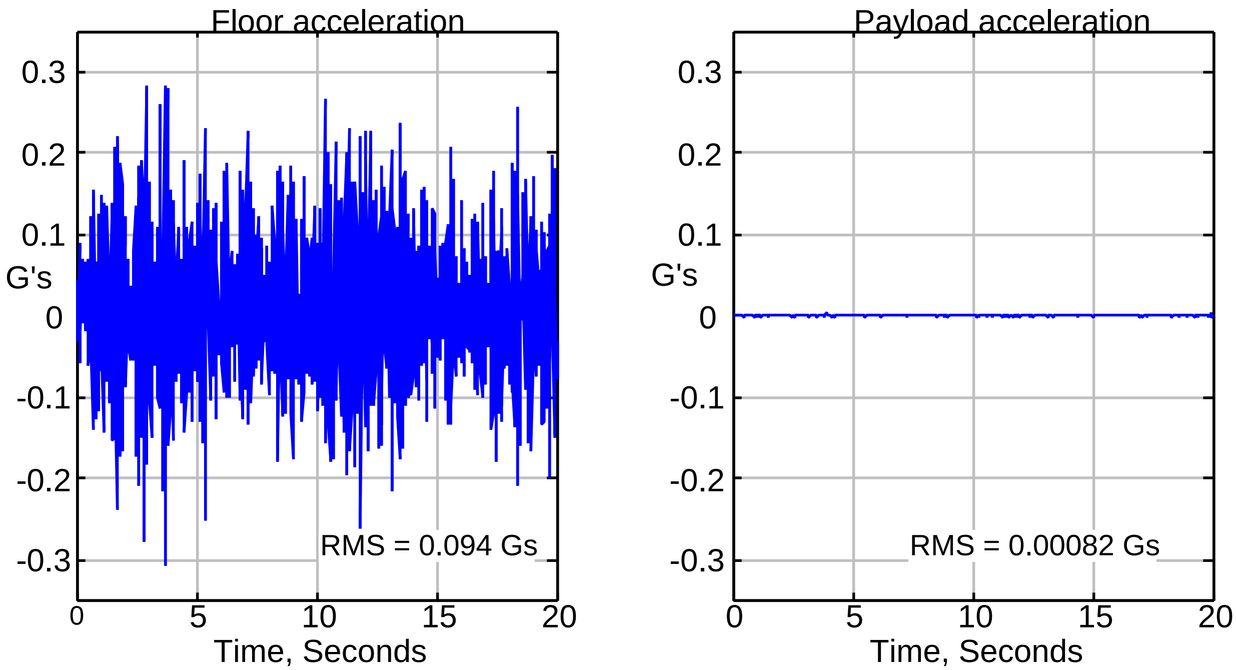


Figure 7. Measured acceleration on floor (left) and on isolated payload (right).

4. CONCLUSION

The combination of semi-passive pneumatic and mechanical suspension has demonstrated the ability to produce high levels of isolation with suspension frequencies below 1 Hz. Vertical suspension frequency can be pushed virtually as low as necessary simply by increasing the accumulator tank size. The moving-piston air spring produces highly linear behavior over a wide range of inputs, including displacements on the order of one inch.

The high-pass character of the pneumatic stiffness implies that the vertical suspension resonance will be well damped without degrading isolation at higher frequencies. Without active augmentation the system demonstrates a combination of 40 dB/decade rolloff plus high damping of vertical suspension resonance. Active augmentation can improve this further as well as providing the means for damping of horizontal suspension resonances.

The voice coil actuators have demonstrated force capacity and bandwidth sufficient to offset maneuvering loads up to several tenths of a G. Current efforts are focused on using the magnetic actuators to maintain dynamic alignment of the entire bench assembly relative to an external target.

5. ACKNOWLEDGEMENT

This work was supported by the Air Force Research Laboratory, Kirtland Air Force Base, NM through the Small Business Innovation Research program. Mr. Gene Fosness was the technical monitor.

REFERENCES

1. Harris, C. (Editor)., *Shock and Vibration Handbook*, Fourth Ed., Section 32, McGraw-Hill.
2. Kienholz, D.A., "Simulation of the Zero-Gravity Environment for Dynamic Testing of Structures," Institute of Environmental Sciences, 19th Space Simulation Symposium, October 28-31, 1996, Baltimore, MD.